

Thermodynamic analysis of a gamma type Stirling engine in non-ideal adiabatic conditions

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ABSTRACT

In this study, a thermodynamic analysis of a gamma type Stirling engine is performed by using a quasi steady flow model based on Urieli and Berchowitz's works. The Stirling engine analysis is performed for five principal fields: compression room, expansion room, cooler, heater and regenerator. The conservation law of the mass and the energy equations are derived for the related sections. A FORTRAN code is developed to solve the derived equations for all process parameters like pressure, temperature, mass flow, dissipation and convection losses for the different spaces (compression space, cooler, regenerator, heater and expansion space) as a function of the crank angle. The developed model gave more precise results for the pressure profile than the models available in the literature.

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1. Introduction

Growing demand for energy and increasing environmental problems have intensified the research on clean energy and new engines that will reduce pollution. The performance of Stirling engines meets the demands of the efficient use of energy and environmental security and therefore they are the subject of current interest in scientific research and industrial applications. Hence, the development and investigation of Stirling engines has become an important aspect for many scientific institutes and commercial companies [1–5]. The Stirling engine is both practically and theoretically a significant device; its practical virtue is that it is simple, reliable and safe which has been recognized for a full century since its invention.

Stirling engines have a long history since the beginning of the 19th century. In 1816, this type of machine was invented by Robert Stirling. He patented his valveless external combustion engine incorporating a regenerative heat exchanger [6]. The Stirling engine always enchanted engineers and physicists because of the theoretical potential to approach the Carnot efficiency [7–11]. However, the use of this engine was not successful due to the relatively poor material technologies available [12]. The development of internal

combustion engines and electric motors surpassed the Stirling engines during the 20th century.

Stirling engines can be used with various energy sources such as biomass or solar radiation and can achieve a high efficiency combined with low environmental risks. As a consequence of this behaviour, the interest in Stirling engines is arising again.

The Stirling motor is an engine with external heat input, which converts heat into mechanical work theoretically at Carnot efficiency. The theoretical Stirling cycle consists of two main changes of state at constant volume and at constant temperature. With respect to the theoretical cycle a Stirling engine has five sections: expansion and compression space, regenerator, cooler and heater. The working fluid remains within the machine in contrast to the Diesel and Otto cycle and flows between the compression cylinder and the expansion cylinder. The working gas transports energy from the high temperature heat source to the low temperature heat sink. Hence, work is obtained during one complete cycle, since the energy needed for compression is less than that emitted during expansion.

The first analysis of the Stirling engine was performed almost 50 years after its invention by Schmidt [13]. Schmidt assumed that the compression and cooler spaces are kept at constant and low temperature and the expansion and heater spaces remain at constant and high temperature. In addition, he assumed a sinusoidal dependency of the volume of the crank angle and calculated the pressure within the engine as a function of the volume.

However, working spaces of real engines have a tendency to be adiabatic rather than isothermal. Analysis with non-isothermal

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Nomenclature		VSE	swept volume of the expansion space (m ³)
A	area (m ²)	W	work (J)
c _{mr}	heat capacity of regenerator matrix (J/kg)	Subscripts c compression space e expansion space h heater i inlet k cooler o outlet r regenerator w wall Greek α crank angle κ specific heat ratio λ connecting rod ratio φ phase shift angle η efficiency	
c _p	specific heat at constant pressure (J/kg K)		
c _v	specific heat at constant volume (J/kg K)		
d	diameter (m)		
DISS	dissipation derivate (J/CAD)		
H	heat transfer coefficient (W/m ² K)		
M	fluid mass (kg)		
P	pressure (N/m ²)		
P	power (W)		
Q	heat (J)		
R	gas constant (J/kg K)		
T	temperature (K)		
V	volume (m ³)		
VDC	dead volume of compression space (m ³)		
VDE	dead volume of expansion space (m ³)		
VSC	swept volume of the compression space (m ³)		

working spaces was first performed by Finkelstein in 1960 [14]. He assumed that the heat transfer within the working spaces occurs by convection. Since the variation of gas temperatures leads to temperature discontinuities, Finkelstein introduced the concept of conditional temperatures depending on the direction of the gas flow. Considering adiabatic compression and expansion spaces, the temperature within the working spaces varies during the whole cycle. The different cells of the engine are assumed as control volumes and the analysis is performed using the equation of state as well as energy and mass conservation laws. The adiabatic analysis, which includes computer codes, has been reconsidered by Lee [15] and Urieli and Berchowitz [16]. Urieli also analysed different engines using the ideal adiabatic model. In addition, computer simulations of Stirling engines with five working spaces were carried out by Martini [17]. He assumed that the working gas temperature in the compression and expansion space are uniform; hot and cold spaces are kept at constant temperature and the regenerator has a linear temperature profile.

Since the isothermal and adiabatic analysis is applied to the same engine with identical assumptions, the results may not be too different. Both calculation models do not consider any losses within the machine, which results in high efficiency. Real engines, however, have many losses, such as flow losses, gas leakage, heat losses, discontinuous piston motion or dead volume effects. In real engines, the efficiency obtained differs from the computed values. Studies presented by Urieli and Berchowitz [16] indicate that the quasi-steady flow model is more sophisticated than steady flow models. This quasi-steady flow model considers non-ideal heat exchangers, because heat exchangers have the biggest influence on the performance of the machine.

In practice, there are three types of Stirling engines: high temperature difference machines, low temperature difference machines and cryo-coolers [18]. In addition, Stirling engines can be distinguished by the configuration of the working spaces and by the kind of mechanical piston control. Distinguishing them by working space configuration there are three types of Stirling cycle engines: α-type machines, β-type machines and γ-type machines. In this study, the gamma type Stirling engine is analysed using the quasi-steady flow model.

2. Model

For this calculation model the equation set of Urieli and Berchowitz is used. Fig. 1 shows the model that the analysis is based

on. Some modifications are done on the equation set in order to make it more reasonable, especially for calculating different pressures for each cell.

A Stirling engine consists of mainly five spaces: compression, cooler, regenerator, heater and expansion space. Since the regenerator is divided into two parts, there are six parts and five interfaces, displayed in Fig. 1. At the beginning of the cycle, it is assumed that the working gas flows from the compression space to the expansion space. The temperature variation of the Stirling engine as a function of length is shown on Fig. 1.

The boundary temperature between two cells depends on the direction of the flow. Therefore these temperatures are determined to be conditional. Referring to Fig. 1 the different regenerator temperatures are combined with linear relations. The regenerator is divided into two cells r₁ and r₂, each cell being associated with its constant temperature for the whole regenerator cell T_{r1} and T_{r2} and corresponding effective matrix temperature T_{wr1} and T_{wr2}. T_{rk}, T_{rr} and T_{rh} are calculated as follows:

$$T_{rk} = \frac{3 \cdot T_{r1} - T_{r2}}{2} \quad (1)$$

$$T_{rr} = \frac{T_{r1} + T_{r2}}{2} \quad (2)$$

$$T_{rh} = \frac{3 \cdot T_{r2} - T_{r1}}{2} \quad (3)$$

The boundary temperatures between two neighbouring cells depend on the direction of flow. The following equations show these relations for the model mentioned in Fig. 2. According to this, the pressure can be calculated analogously.

$$\text{if } m_{ck} > 0 \text{ then } T_{ck} = T_c \text{ else } T_{ck} = T_k \quad (4)$$

$$\text{if } m_{kr} > 0 \text{ then } T_{kr} = T_k \text{ else } T_{kr} = T_{rk} \quad (5)$$

$$\text{if } m_{rh} > 0 \text{ then } T_{rh} = T_{rh} \text{ else } T_{rh} = T_h \quad (6)$$

$$\text{if } m_{he} > 0 \text{ then } T_{he} = T_h \text{ else } T_{he} = T_e \quad (7)$$

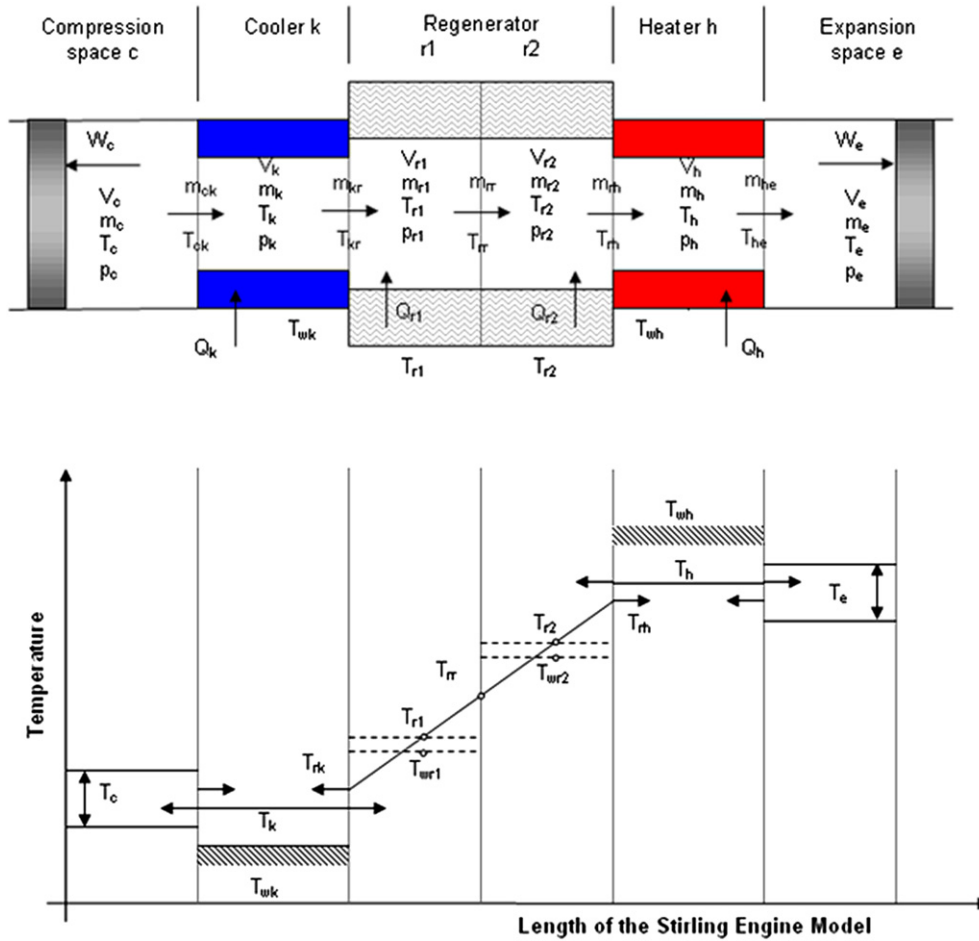


Fig 1. Calculation model.

The heat flux within the heat exchangers can be calculated by using the equations for the convective heat transfer, which requires the value of the heat transfer coefficients h_k , h_{r1} , h_{r2} and h_h . In addition, it is very important to take care of the right units for the variables in these equations. Thus, the factor dz is added to compensate for the unit of the heat derivative ($J/^\circ$ instead of W):

$$DQ = \frac{h \cdot A \cdot (T_w - T)}{dz} \quad (8)$$

Since the heat transfer between gas and wall is considered in this analysis, the determination of the exact wall temperature is

very important. Heater and cooler wall temperatures T_{wk} and T_{wh} are assumed to be constant. This assumption however cannot be made for the regenerator wall temperature, which depends on the energy stored in the matrix and is described with the following equation:

$$DT = \frac{DQ_r}{c_{mr}} \quad (9)$$

where c_{mr} is the heat capacity of the regenerator matrix and shows how much energy is needed to heat up the matrix.

The formulation of the equation for the pressure and its derivative is based on the energy balance for several cells. A generalised cell, shown in Fig. 2, may either be reduced to a working space cell or a heat exchanger cell. Enthalpy is transported into the cell by means of the mass flow $\dot{m}_i$ and temperature T_i , and transported out of the cell by the mass flow $\dot{m}_o$ and the temperature T_o .

The energy balance for a generalised cell as shown in Fig. 2 is as follows:

$$DW + DQ + DISS + (c_p \cdot T_i \cdot \dot{m}_i - c_p \cdot T_o \cdot \dot{m}_o) = c_v \cdot D(m \cdot T) \quad (10)$$

Eq. (10) is the well-known energy equation in which kinetic and potential energy terms have been neglected.

The assumption is made that the pressure derivative is the same in all cells. In this step the pressure drop caused by friction with its very small value in contrast to the pressure derivative is neglected. Assuming an ideal gas and using Eqs. (11) and (12), Eq. (10) is modified and written as Eq. (13):

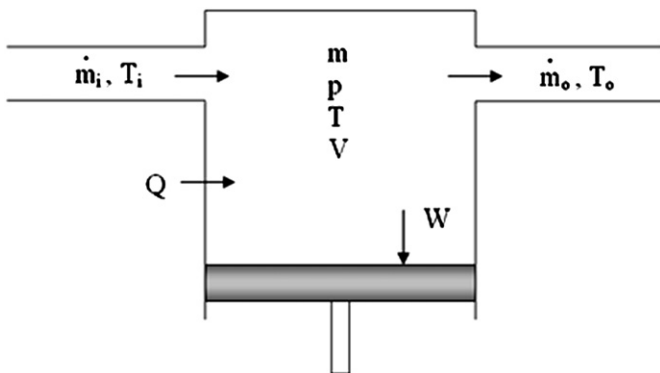


Fig 2. Generalized cell.

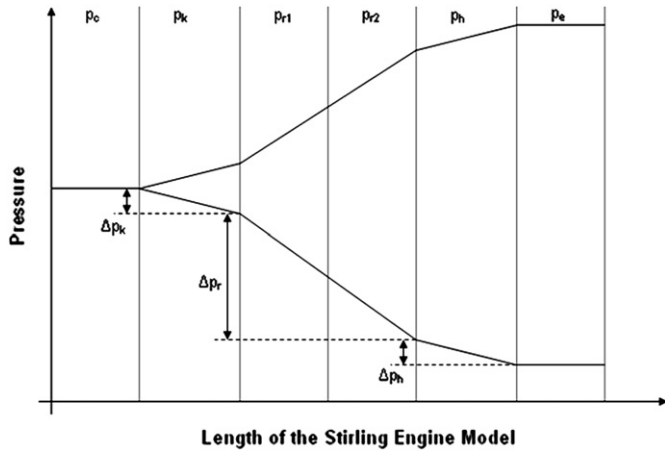


Fig 3. Calculation model of the pressure.

$$D(m \cdot T) = \frac{p \cdot DV}{R} + \frac{V \cdot Dp}{R} \quad (11)$$

$$W = -p \cdot DV \quad (12)$$

$$DQ + DISS + (c_p \cdot T_i \cdot \dot{m}_i - c_p \cdot T_o \cdot \dot{m}_o) = \frac{c_p}{R} \cdot p \cdot DV + \frac{c_v}{R} \cdot V \cdot Dp \quad (13)$$



Fig 4. The biomass Stirling engine.

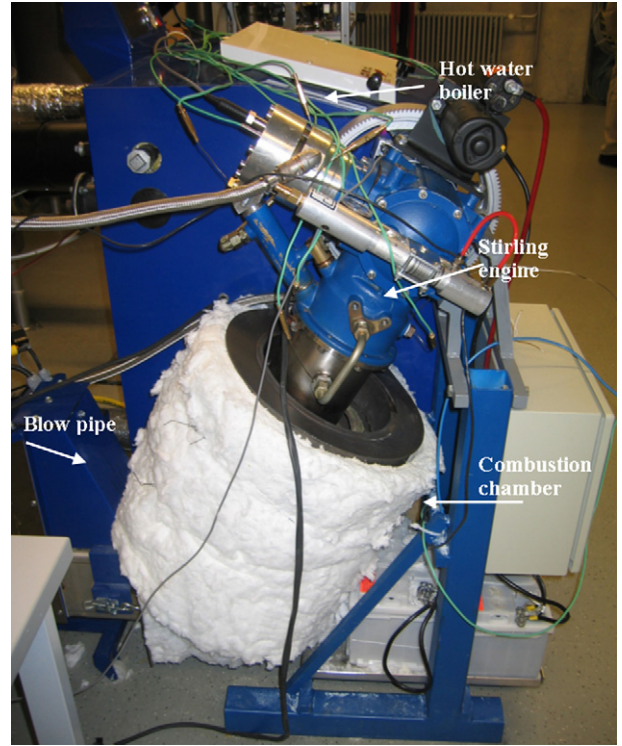


Fig 5. Stirling engine integrated in the heating system.

Applying this equation to each section of the Stirling engine and summing up all equations, a general energy balance for the whole system is built up:

$$DQ + DISS = -\frac{c_p}{R} \cdot DW + \frac{c_v}{R} \cdot V \cdot Dp \quad (14)$$

DQ is the total derivative of the heat flux and DISS the dissipation of the whole system; DW is the total derivative of the work which is emitted by the engine. Rearranging Eq. (14), Dp can be calculated as follows:

$$Dp = \frac{R \cdot (DQ + DISS) + c_p \cdot DW}{c_v \cdot V} \quad (15)$$

Presuming that the pressure derivative is constant in all the cells, the different pressures of each part in the system can be calculated. As an example the pressure derivative of the compression space is shown below:

$$Dp_c = \frac{R \cdot (DQ + DISS) + c_p \cdot DW}{c_v \cdot V} \quad (16)$$

Table 1
Technical data of the biomass engine

Motor type	γ-type Stirling
Motor output power	1000 W _{peak}
Motor rated speed	700 min ⁻¹
Generator type	3 phase a.c. motor, permanent magnetic
Generator output power	820 W _{peak}
Boiler	SBS Axiom
Boiler thermal performance	21.4 kW
Blow pipe	SBS Janfire flex-a
Fired by	Wooden pellets
Motor weight	43 kg
Working gas	Nitrogen
Working gas preload pressure	7.5 bar
Heater temperature	~1100 °C
Coolant temperature	~50 °C
Manufacturer	EPAS GmbH, Germany

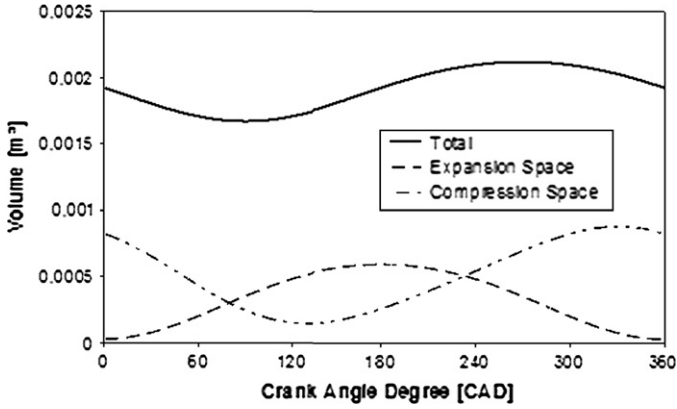


Fig 6. Variation of the compression, expansion and total volume.

Fig. 3 shows the pressure distribution within the system. The pressure drop Δp is calculated taking into account the friction between wall and fluid. Since the pressure drop depends on the flow direction, the pressure in the different cells depends also on the flow direction and should be calculated according to Fig. 3. If the working gas flow starts at the expansion space, the pressure decreases due to the pressure drop in the direction of the compression space and vice versa.

For the calculation of the mass in each cell, its derivative and the mass flow across each cross section, the mass derivative Dm_c in the compression space is calculated. Respecting continuity considerations Dm_c can be calculated as shown below:

$$Dm_c = -\dot{m}_{ck} \quad (17)$$

Eq. (17) shows that the mass flow across the cross section between the compression space and the cooler is directly proportional to the change of the mass in the compression space. Applying the energy equation to the compression space and implementing the ideal gas relation, the mass flow rate Dm_c can be calculated as follows:

$$Dm_c = \frac{p_c \cdot DV_c + (1/\kappa) \cdot V_c \cdot Dp_c}{R \cdot T_{ck}} \quad (18)$$

Using Dm_c and the energy balance for the different spaces, the other mass flow rates in the cross sections between the cells can be determined with Eq. (19):

$$\dot{m}_o = \frac{c_p \cdot T_i \cdot \dot{m}_i + DQ + DISS - \frac{c_v}{R} \cdot V \cdot Dp}{c_p \cdot T_o} \quad (19)$$

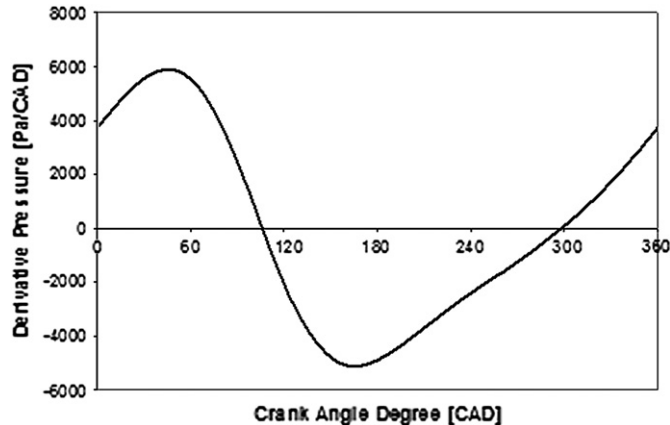


Fig 7. Derivative pressure variation.

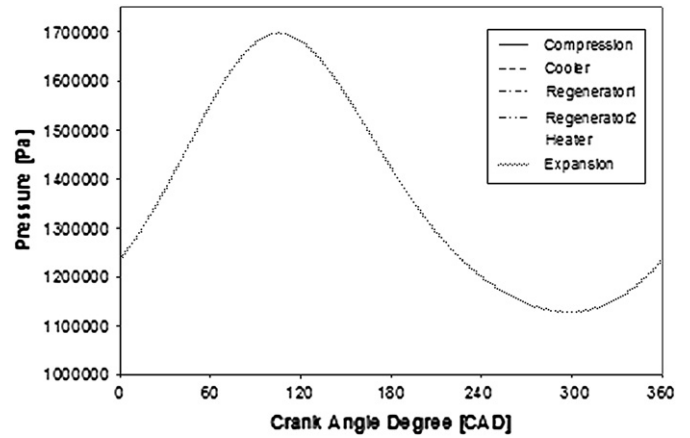


Fig 8. Pressure variation for compression, cooler, regenerator, heater and expansion space.

The last step in the derivation of the equation set is the determination of the cell temperatures. This is simply done with the equation of state, because all values needed were calculated before:

$$T = \frac{p \cdot V}{R \cdot m} \quad (20)$$

A consistent set of initial conditions has to be chosen for the system variables and the set of equations is subsequently integrated through several complete cycles until cyclic steady state has been attained. At each integration increment, the values of the heat transfer coefficient h and pressure drop Δp are evaluated in accordance with the methods presented before.

The time needed for convergence is mainly dependent on the thermal capacitance of the system – in particular that of the regenerator matrix.

2.1. Stirling engine losses

Although Stirling engines seem to have a high thermal efficiency, they have several sources of losses reducing the power output. The flow loss is one of the important losses of the Stirling engine. It occurs in the heat exchanger parts of the machine, especially within the regenerator, where the share of flow losses is about 90% of the total flow loss due to a small cross section.

Further losses are heat losses especially on the hot surfaces of the Stirling engine, which have to be considered in the analysis. The

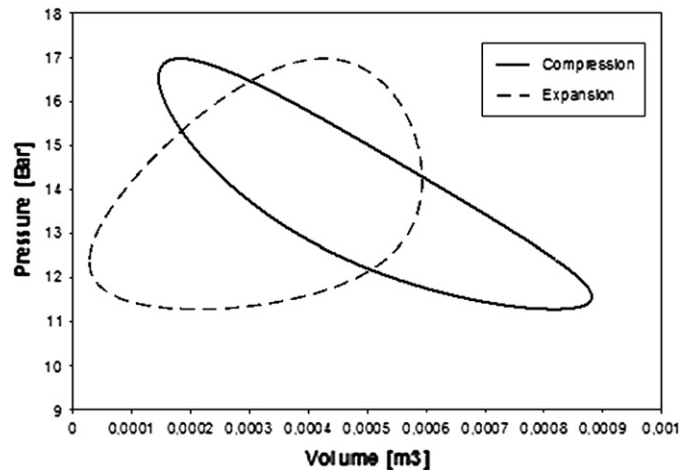


Fig 9. p - V diagram for compression and expansion space.

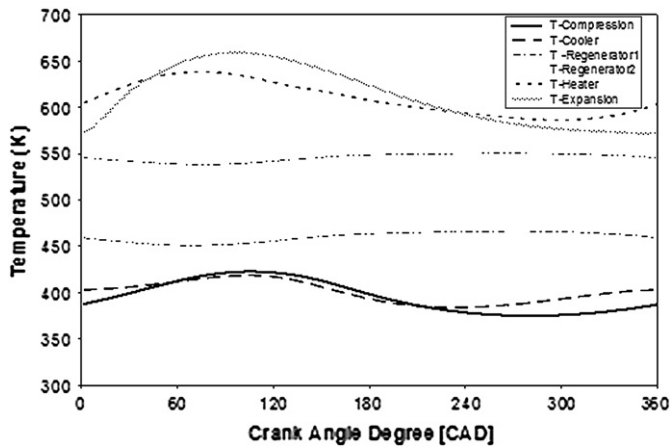


Fig. 10. Temperature variation of the working gas.

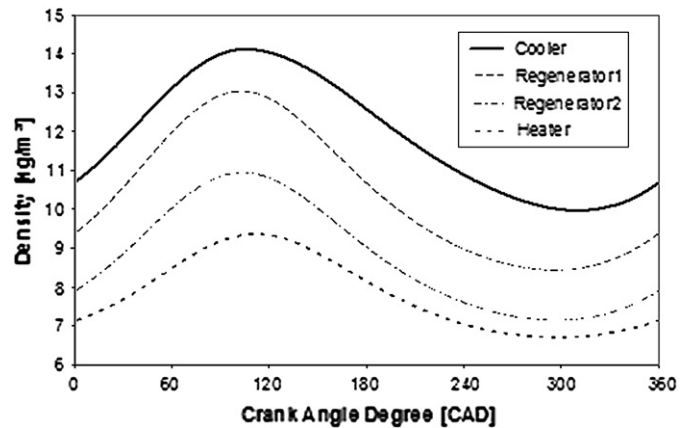


Fig. 12. Variation of the density as function of the crank angle.

losses derived by Makhkamov [19] become predominant factors when the engine speed increases since they are proportional to the square of speed [1]. Helium and hydrogen with their light molecular weight can be used as working fluids to reduce these losses.

The unswept volumes (dead volumes in a Stirling engine) should be kept zero in theory although it can reach up to 50% of the total engine internal gas volume in most of the real engines. In addition the effects of dead volumes within the motor on the engine efficiency have to be taken into account. These dead volumes cause a lower pressure level and also a decrease of the total efficiency. Feng Wu [20] developed a criteria for optimization of dead volume. The amount of dead volume is required to hold the necessary heat exchangers and to contain sufficient heat transfer surfaces. Dead space, which can be altered during operation of the engine to control power output, reduces the power output of the engine but has conflicting effects on the efficiency, depending on the location of the dead space [1].

Further losses are caused by working gas leakage, which requires a sealing of the motor. Particularly the working piston has to be sealed, because on this part the biggest amount of performance can be lost. In this study the pressure drop due to friction and heat losses are considered.

3. Experimental setup

This system, the EPAS BM1000, with a maximum electrical performance of 820 W is a combination of a biomass Stirling engine

for the generation of electrical current and a heating and hot water boiler as seen in Figs. 4 and 5. The main components of the system are the blowpipe, the hot water boiler, the Stirling engine, water supply with pumps for the Stirling engine and the hot water boiler and electric installations. For the production of electrical power, the Stirling engine is mounted in a combustion chamber between the blowpipe and the hot water boiler. The Stirling engine has an electric power of 1 kW and a thermal output of 23 kW. Technical data of the biomass engine are given in Table 1. The result of the combined heat and power production is the very high efficiency of the system.

The rated electrical performance of 820 W can only be reached if the blow pipe produces its maximum temperature and output. If the system operates in part load condition the Stirling engine adapts its performance. For optimum operational conditions the blow pipe has to produce at least 20 kW heat. This is the reason for the modification mentioned before. As a working gas, nitrogen is added to the engine with a preload pressure of 6.5 bar. If a pressure loss occurs the system has to be filled up in the cold state. At the University of Applied Sciences in Regensburg this is done with the compressed air system and a nitrogen filter membrane. As is shown in Section 5 different parts of the Stirling engine have to be cooled. To achieve good performances a volume flow of 10–20 l/min cooling water with a temperature of 45–60 °C has to be provided. The Stirling engine can be started if a combustion gas temperature of approximately 600 °C is reached. The starting procedure has to be initiated by a button on the switch box.

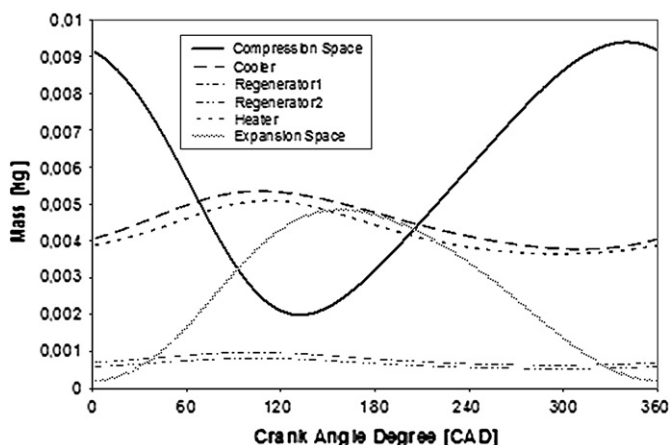


Fig. 11. Variation of the working gas mass.

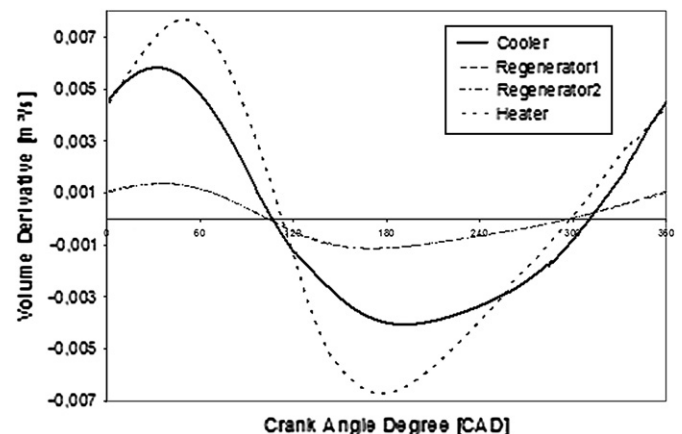


Fig. 13. Derivative volume as function of the crank angle.

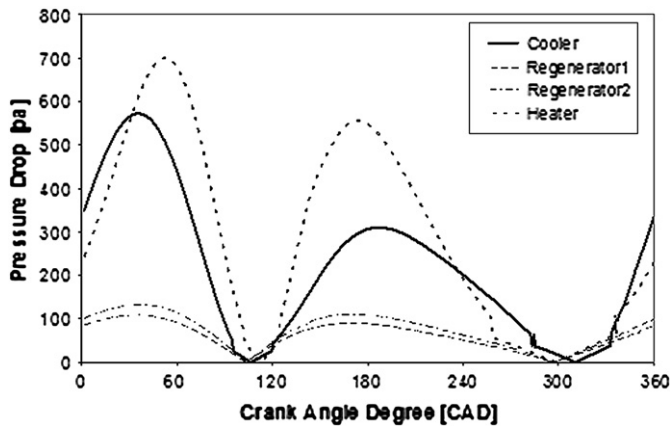


Fig. 14. Pressure drop within the heat exchanger cells.

4. Results

The volumes of the compression and expansion space as a function of the crank angle have been calculated using the following Eqs. (21) and (22):

$$V_e = \frac{VSE}{2} \cdot \left[1 - \cos(\alpha) + \frac{1}{\lambda_e} \cdot \left(1 - \sqrt{1 - \lambda_e^2 \cdot \sin^2(\alpha)} \right) \right] + VDE \quad (21)$$

$$V_c = \frac{VSE}{2} \cdot \left[1 + \cos(\alpha) - \frac{1}{\lambda_e} \cdot \left(1 - \sqrt{1 - \lambda_e^2 \cdot \sin^2(\alpha)} \right) \right] + \frac{VSC}{2} \cdot \left[1 - \cos(\alpha - \phi) + \frac{1}{\lambda_c} \cdot \left(1 - \sqrt{1 - \lambda_c^2 \cdot \sin^2(\alpha - \phi)} \right) \right] + VDC \quad (22)$$

Fig. 6 shows the variation of the compression volume, expansion volume and total volume as a function of the crank angle. The compression volume reaches its maximum value when the cycle has not begun yet, because the flow direction of the working gas has been chosen from compression space to the expansion space. Starting in the compression space, the working gas is pumped through the cooler, the regenerator and the heater to the expansion space, successively. As the total volume V increases, the volume of the compression space V_c increases too, whereas the volume of the expansion space V_e decreases. After the working gas has been expanded in the expansion space, the flow turns back to the compression space.

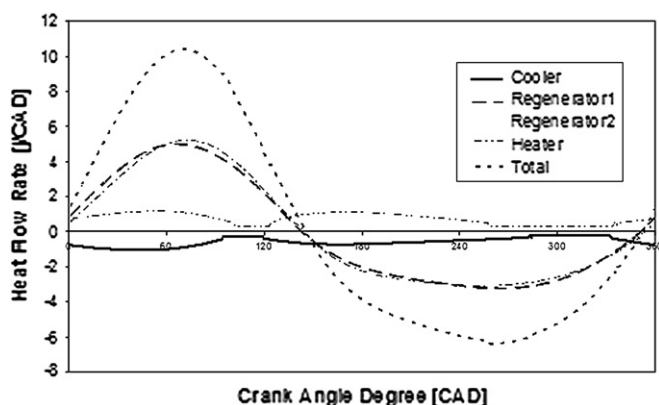


Fig. 15. Heat transferred within the different cells.

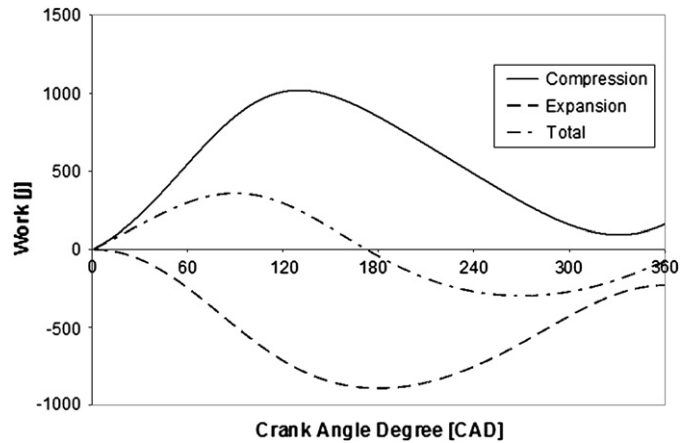


Fig. 16. Variation of work obtained during the cycle as function of the crank angle.

The derivative pressure is displayed in Fig. 7. Fig. 8 shows the pressure as a function of the crank angle. It can be seen in Fig. 8, that there is nearly no difference in pressure within the different parts of the engine.

The pressure of the engine cells is calculated by considering pressure drops. The work during compression and expansion is shown in form of a p - V diagram in Fig. 9.

Fig. 10 shows the variation of the working gas temperature with the crank angle. The temperature within the compression and expansion space increases due to the decreasing total volume before the total volume starts to increase again. Similarly, the temperature in the heater and cooler increases due to heating until the working gas has not expanded. The gas temperature within the two regenerator cells is nearly constant during the whole cycle. As the total volume increases, the cooling occurs due to the expansion of the gas and the temperature of the cooler, heater, compression and expansion space decreases when the flow turns back. Determination of the mass of working gas is of critical importance in the calculation of friction factor and Nusselt number. Mass flow rates have been calculated using Eq. (20). Figs. 11 and 12 show the variations of mass and density of working gas, respectively. Derivative volume as a function of the crank angle is shown in Fig. 13. Fig. 14 shows the pressure drop within the heat exchanger cells during the cycle. Heat transferred within the different cells has been presented in Fig. 15.

The energy balance was performed by calculation of the overall heat and total work. The total work was calculated using pressure

Table 2
Values for the modified engine

Parameter	Original	Modified	Unit	%
Gas	Air	He	–	–
ϕ	9.00E+01	7.50E+01	°	–1.67E+01
VSC	4.51E–04	4.00E–04	m ³	–1.13E+01
VSE	5.65E–04	7.00E–04	m ³	2.39E+01
m_{gas}	1.86E–02	2.51E–03	kg	–8.65E+01
A_k	1.03E–03	1.62E–03	m ²	5.73E+01
A_{wk}	9.45E–02	2.38E–01	m ²	1.52E+02
d_k	4.00E–03	2.50E–03	m	–3.75E+01
V_k	3.79E–04	4.38E–04	m ³	1.56E+01
T_{wk}	2.95E+02	2.95E+02	K	0.00E+00
T_{wh}	1.18E+03	1.38E+03	K	1.70E+01
p_c max	1.72E+04	1.57E+04	bar	–8.69E+00
T_k min	4.31E+02	2.86E+02	K	–3.37E+01
T_h max	6.02E+02	1.14E+03	K	8.98E+01
Q_k	–2.29E+02	–1.33E+02	J	–4.20E+01
Q_h	2.63E+02	3.12E+02	J	1.90E+01
W	–3.50E+01	–1.81E+02	J	4.16E+02
P	4.08E+02	2.11E+03	W	4.16E+02
η	1.33E–01	5.78E–01	–	3.34E+02

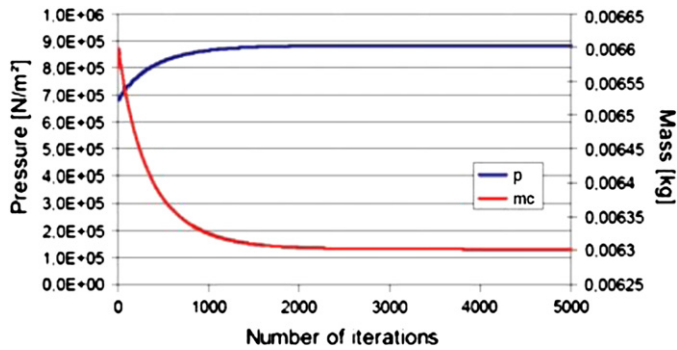


Fig 17. Convergence behavior.

and volume as a function of the crank angle. The variation of work which is obtained during the cycle as function of the crank angle is presented Fig. 16.

5. Application

In this section, an application [21] based on our model by using the same experimental setup will be given. Table 2 shows the boundary conditions and the results for the original and the modified engine. It can be clearly seen, that with the calculation method developed, parameters can be found with which the theoretical performance can be raised by a factor of 4 and the efficiency can be raised by a factor of 3.

Fig. 17 shows the convergence behaviour of the compression space pressure and mass. It takes more than 2000 iterations to reach cyclic steady state, because the regenerator matrix takes many iterations to reach its final wall temperature. All the values are shown for the same crank angle (360°).

Fig. 18 shows the p - V diagrams as an indicator of the work done by the circle for both engines (biomass and solar Stirling engine). It describes the p - V diagram of the whole process with the compression space pressure and the total volume.

Fig. 19 shows the temperature profile of the different cells over the whole cycle in one diagram for the biomass engine. The green and red lines mark the minimum and maximum temperatures. It can be interpreted that the gas temperatures of the cooler and heater are far away from the wall temperatures. This shows that there are huge heat convection losses in the heater sections diminishing the efficiency enormously.

6. Conclusions

In this work, a numerical method is demonstrated for obtaining a thermodynamic analysis of a gamma type Stirling engine. The analysis provides the necessary data for the comparison of several aspects of the Stirling engine. The method has been applied to maximize the power output and the corresponding thermal

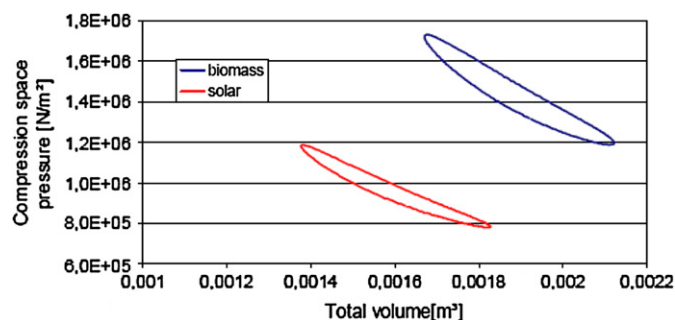
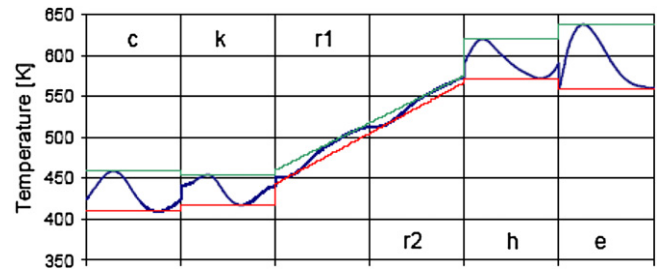
Fig 18. p - V diagram of the whole process.

Fig 19. Temperature for the different cells.

efficiency of a gamma type Stirling engine. The analysis is performed using quasi steady flow model based on Urieli's and Berchowitz's works. Five main components of the Stirling engine that are compression and expansion space, cooler, heater and regenerator, are analysed separately. The equations like the conservation law of mass and energy are derived for each volume. A computer code is developed and the equations are solved by using the code. The variations of mass, pressure and temperature are calculated. This work provides data that will contribute to the improvement of the Stirling engine. Finally, the thermal efficiency of the Stirling engine has been calculated with a value of 25%. A future step of this analysis is to validate this model against experimental data which will be obtained in the experimental setup.

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